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Axial piston compressor, especially a compressor for the air-conditioning system of a motor vehicle

Description

The invention relates to an axial piston compressor, especially to compressors for motor vehicle air-conditioning systems, having a housing and, for drawing in and compressing a coolant, a compressor unit arranged in the housing and driven by means of a drive shaft, the compressor unit comprising pistons, which move axially back and forth in a cylinder block, and a tilt plate (swash ring, tilt ring or wobble plate) which drives the pistons and rotates together with the drive shaft.

An axial piston compressor of such a kind is known, for example, from DE 197 49 727 A1. That compressor comprises a housing in which, in a circular arrangement, a plurality of axial pistons are arranged around a rotating drive shaft. The drive force is transmitted from the drive shaft, by way of a member for conjoint movement, to an annular tilt plate and in turn, from there, to the pistons displaceable in translation parallel to the drive shaft. The annular tilt plate is pivotally mounted on a sleeve which is mounted on the drive shaft so as to be axially displaceable. In the sleeve there is provided an elongate hole, through which the mentioned member for conjoint movement engages. Consequently, the capability of the sleeve for axial movement on the drive shaft is limited by the dimensions of the elongate hole. Assembly is carried out by passing the member for conjoint movement through the elongate hole. The drive shaft, member for conjoint movement, sliding sleeve and tilt plate are arranged in a so-called drive mechanism chamber, in which gaseous working medium of the compressor is present at a particular pressure. The delivery volume and consequently the delivery output of the compressor are dependent on the pressure ratio between the suction side and delivery side of the pistons or correspondingly dependent on the pressures in the cylinders on the one hand and in the drive mechanism chamber on the other hand.

A somewhat different kind of construction of an axial piston compressor is described, for example, in DE 198 39 914 A1. The tilt plate is in the form of a wobble plate, there being arranged between the wobble plate and the pistons a non-rotating take-up plate mounted opposite the wobble plate.

Reference is made, furthermore, to the following prior art:

- DE 2 524 148
- US 4 815 358
- US 4 836 090
- US 4 077 269
- US 5 105 728

In the case of the compressors described in those publications, the purpose is, *inter alia*, to take measures to prevent or reduce drive mechanism imbalance in use. Otherwise the known arrangements have in common the fact that the rotating components are of relatively large and, consequently, heavy construction compared to the parts moved in translation, namely the pistons, piston rod etc.. Furthermore, the known arrangements have in common the fact that the actual tilt plate apparatus is acted upon by an additional plate by means of a suitable coupling mechanism. The several rotating components are intended to bring about a righting moment of the tilt plate apparatus in the direction of minimum piston stroke, which has an influence on the regulation behaviour.

The mentioned arrangements are all relatively complicated, expensive and of low compactness and for that reason they are unsuitable for the compressors required nowadays by the automobile industry for air-conditioning systems.

Also in the case of mass-produced compressors as are used in motor vehicles, it is an objective that the components moved (especially their mass) should be suitably dimensioned in order to achieve a desired regulation behaviour. The compressor 6SEU 12C mass-produced by DENSO has, for example, a drive mechanism having the following masses relevant to the regulation behaviour:

Component	Number	Mass of component [g]	Total mass [g]
Pistons	6	41	246
Sliding block	12	5	60
Masses moved in translation			**306 g**
Swash plate	1	391	391
Guide pins	2	20	40
Masses moved in rotation			**431 g**

From the above-mentioned figures it can be seen that a considerable component mass is provided for the parts moved in rotation. By that means an attempt is made to produce a sufficient counter-force or counter-moment relative to the masses moved in translation. The same basic idea also underlies DE 198 39 914 A1, in which indeed the rotating mass of the tilt plate or of the pivotal part thereof is so dimensioned that the centrifugal forces occurring on rotation of the drive plate are sufficient to counteract the pivotal movement of the tilt plate to provide deliberate regulation and consequently to influence, namely to reduce or to limit, or especially to keep constant, the piston stroke and accordingly the quantity delivered.

The influences acting as moments about the tilt centre of a tilt plate apparatus are, in detail, the following moments, the direction of the moments being given in brackets, with (-) denoting down-regulation (in the direction of minimum stroke) and (+) denoting up-regulation (in the direction of maximum stroke):

- moment due to gas forces in the cylinder spaces (+)
- moment due to gas forces from the drive mechanism chamber (-)
- moment due to a restoring spring (-)
- moment due to an advancing spring (+)
- moment due to rotating masses (-); including moment due to location of centre of gravity (for example, tilt plate: tilt location ≠ mass centre of gravity) : can be (+) or (-)
- moment due to masses moved in translation (+)

In relation to the mentioned 6SEU 12C compressor of DENSO, which represents the typical constructional form of a tilt plate compressor, it is to be noted that the mass of such a tilt plate cannot be increased at will in order to modify the regulation behaviour

accordingly. This is due to the fact that, in the case of the compressors of the described kind, the mass centre of gravity of the tilt plate is generally a substantial distance away from the tilt-providing articulation of the tilt plate. The basic justification for such an arrangement is that the tilt plate, in addition to its own guideway on the drive shaft, has to be coupled to the drive shaft or a component connected to the drive shaft by way of a positioning mechanism.

The mentioned distance between the centre of gravity of the tilt plate and the tilt-providing articulation thereof results in imbalance of the drive mechanism, especially in dependence upon the tilt plate tilting angle (the centre of gravity moves "in the manner of a swing" beneath the tilt-providing articulation), and in the worst case results in an up-regulating characteristic (so-called "location of centre of gravity").

Accordingly, in the case of the compressors according to the prior art, and indeed according to both the published and the actually practised prior art, a compromise has to be reached so that a predetermined mass of the tilt plate is made available in order to produce a counter-moment to the masses moved in translation; on the other hand, however, the mass of the tilt plate must not be over-dimensioned because then the imbalance of the drive mechanism would be excessive. Otherwise, when a tilt plate is constructed in the form of a tilt ring, increasing the mass thereof is restricted by the available space.

In order to address that problem it has also already been proposed that the pistons, that is to say the masses moved in translation, should be constructed as sparingly, that is to say as lightly, as possible, for example using aluminium or other materials of relatively low specific density. In that respect it has also been proposed to use hollow pistons.

Reference is made furthermore to the compressor according to EP 0 809 027 A1, which relates to a particular arrangement of the coupling mechanism between the drive shaft and tilt plate apparatus. The coupling mechanism is designed for high pressure, for example when R744 is used as coolant. Also of importance in this last-mentioned prior art is so-called constant regulation of the delivery quantity. It is proposed that the kinematics of the compressor be so designed that the down-regulating tilting moments acting on the tilt plate should clearly predominate over the up-regulating tilting moments. In this context it should be mentioned that the phrase "delivery quantity" is relatively

imprecise. The delivery quantity could be considered constant if, for example, on doubling the speed of rotation, the tilt angle of the tilt plate apparatus halves. As a result the delivery quantity would be constant in geometric terms. Of course, other parameters will also then have an effect on the delivery quantity when the tilt angle of the tilt plate changes, for example volumetric efficiency, oil throw-off or the like.

For constant regulation of the delivery quantity in the event of changing speeds of rotation, the restoring torque of the tilt plate apparatus is utilised because the tilt plate opposes its angled position because of the dynamic forces at the co-rotating plate part. This process can be aided by the force of a spring so that the increasing quantity delivered in the case of an increase in the speed of rotation is at least partly compensated by restoration of the angled or pivoted position of the tilt plate.

As already mentioned hereinbefore, such a behaviour can in principle be obtained by, for example, integrating an additional mass into the drive mechanism, the inertia of which mass acts on the tilt plate by way of a coupling mechanism. It was also explained that, in the case of compressors as are used today in motor vehicles, the mass of the tilt plate cannot be increased at will without having to accept other disadvantages. This also holds true, especially, for the teaching according to DE 198 39 914 A1 and EP Application No. 99 953 619. The regulation proposed therein using the mass of the rotating components may result in regulation behaviour as a result of which the delivery output is substantially independent of the speed of rotation but this is not necessarily the case. Over-compensation may also be an outcome. The design criteria are very imprecise. The reason for that lies in the fact that the righting moment of the tilt plate is influenced only proportionally by the mass of the rotating components but quadratically by the speed of rotation (ω), which is to say that the quantity delivered can be compensated only in the relatively high speed of rotation range (in this case the dynamics play a part) and for exactly 2 speeds of rotation.

Furthermore, compressors are known, especially mass-produced compressors for R134a, wherein the stroke volume has a tendency to increase solely because of the moments of up-regulating and down-regulating mass forces that come into play. In some cases this has to be compensated by means of appropriate regulatory intervention of the regulating valves used. In the case of relatively new developments, especially for CO_2

compressors, attempts are made at reversing that behaviour. The necessary regulatory intervention can then be reduced or can even be dispensed with.

For a better understanding, the described tilting behaviour due to variation in the speed of rotation is shown in Figs. 2 and 3. Fig. 2 shows the dependence of drive mechanism chamber pressure difference, relative to the suction pressure, set against the tilt angle α or "alpha" of the tilt plate. Calculations were carried out by way of example for the following pressures:

high pressure 120 bar and suction pressure 35 bar.

Also calculated were the speeds of rotation:

600 rpm, 1200 rpm, 2500 rpm, 5000 rpm, 8000 rpm and 11,000 rpm.

In Fig. 2, however, only five of the six plots calculated are to be seen. This is due to the fact that the plots for the speeds of rotation 600 rpm and 1200 rpm lie substantially entirely on top of one another (because of a lack of dynamic); accordingly the "delivered quantity that is independent of the speed of rotation", which is required in the prior art, is rather a wishful notion that cannot be put into practice using the measures described.

Referring to the diagram according to Fig. 2, it can be very clearly seen that plots are obtained which cause the tilt plate to adopt greater tilt angles when the speed of rotation increases. The calculation was based on a tilt ring having a predetermined internal and external diameter and a predetermined height.

Also of relevance are the piston mass, the reference diameter on which the pistons are located, and the number of pistons.

The tilt ring preferably has a mass moment of inertia $J_2 = J\eta$ or $J = m/4\ (r_a^2 + r_i^2 + h^2/3)$ which is greater than 100,000 gmm^2. Preferably, the mass moment of inertia is greater than J = 200,000 - 250,000 gmm^2.

Furthermore, the tilt ring preferably has a mass moment of inertia of $J_3 = J_\zeta = \frac{m}{2}$ $(r_a^2 + r_i^2)$ which is greater than 200,000 gmm^2, preferably about 400,000 - 500,000 gmm^2.

There is described hereinbelow the derivation of the so-called moment of deviation, which governs the tilting of the tilt plate or tilt ring and which, more particularly, in the case shown is solely responsible for the tilting of the tilt plate or tilt ring provided that the mass centre of gravity of the tilt plate or tilt ring is located both at the tilting point and also at the geometric centre-point of the tilt plate or tilt ring. This represents an ideal case of the arrangement that is to be aspired to. For the derivation of the moment of deviation the following very generally applies, with reference to Fig. 13:

$$J_{yz} = -J_1 \cos\alpha_2 \cos\alpha_3 - J_2 \cos\beta_2 \cos\beta_3 - J_3 \cos\gamma_2 \cos\gamma_3$$

$$\left.\begin{aligned} \alpha_1 &= 0 \\ \beta_1 &= 90° \\ \gamma_1 &= 90° \end{aligned}\right\}$$ Direction angles of the x axis relative to the main inertia axes $\xi \cdot \eta \cdot \zeta$

$$\left.\begin{aligned} \alpha_2 &= 90° \\ \beta_2 &= \psi \\ \gamma_2 &= 90° + \psi \end{aligned}\right\}$$ Direction angles of the y axis relative to the main inertia axes $\xi \cdot \eta \cdot \zeta$

$$\left.\begin{aligned} \alpha_3 &= 90° \\ \beta_3 &= 90° - \psi \\ \gamma_3 &= \psi \end{aligned}\right\}$$ Direction angles of the z axis relative to the main inertia axes $\xi \cdot \eta \cdot \zeta$

$$J_2 = J_\eta = \frac{m}{4}\left(r_a^2 + r_i^2 + \frac{h^2}{3}\right)$$

$$J_3 = J_\zeta = \frac{m}{2}\left(r_a^2 + r_i^2\right)$$

(Note: $J_3 \approx 2\, J_2$

Aim: J_{yz} should have a particular magnitude

$J_{yz} \uparrow \} \; J_3 \uparrow J_2$ necessarily increases!)

Moment of deviation

$$J_{yz} = -J_2 \cos\psi \sin\psi + J_3 \cos\psi \sin\psi$$

The following holds true independently of Fig. 13:

Moment due to mass force of the pistons

$$\beta_l = \theta + 2\pi (l-1) \frac{1}{n}$$

$$Zl = R \cdot \omega^2 \tan\alpha \cos\beta_l$$

$$F_{ml} = m_k \cdot z_l$$

$$M(F_{ml}) = m_k \cdot R \cdot \cos\beta_l \cdot z_l$$

$$M_{k,ges} = m_k \cdot R \sum_{l=1}^{n} z_l \cdot \cos\beta_l$$

Moment Msw due to moment of deviation

$$M_{sw} = J_{yz} \cdot \omega^2$$

$$J_{yz} = \{\frac{msw}{2}(r_a^2 + r_i^2) - \frac{msw}{4}(r_a^2 + r_i^2 + \frac{h^2}{3})\} \cos\alpha \sin\alpha$$

$$J_{yz} = \frac{msw}{24} \sin 2\alpha\,(3r_a^2 + 3r_i^2 - h^2)$$

$$M_{SW} \geq M_{k,ges}$$

or

$$[\omega^2 R^2 \cdot m_k \tan\alpha \sum_{l=1}^{n} \cos^2\beta = \omega^2 \frac{msw}{24} \sin 2\alpha\,(3r_a^2 + 3r_i^2 - h^2)]$$

The variables used above have the following meanings:

θ	rotation angle of the shaft (the considerations above and below being made on the basis of $\theta = 0$ for the sake of simplicity)
η	number of pistons
R	distance from piston axis to shaft axis
ω	speed of rotation of shaft

α	tilt angle of tilt ring/tilt plate
mk	mass of a piston including sliding blocks or pair of sliding blocks
mk,ges	mass of all pistons including sliding blocks
msw	mass of tilt ring
ra	external radius of tilt ring
ri	internal radius of tilt ring
h	height of tilt ring
ρ	density of tilt ring
V	volume of tilt ring
βi	angle position of piston i
zi	acceleration of piston i
Fmi	mass force of piston i (including a pair of sliding blocks)
M(Fmi)	moment due to mass force of piston i
Mk,ges	moment due to mass force of all pistons
Msw	moment due to righting moment of tilt ring/tilt plate due to moment of deviation (J_{yz})
J =	f (ρ, r, h) mass moment of inertia

Specifically, Fig. 2 was based on the following tilt plate or swash plate tilt moment determination, wherein α was varied from 0° to 16°:

Determination of swash plate tilt moment

theta	0	0,00	[°]
n (p)	7		-
R	29		[mm]
n	2500		[1/min]
alpha	16	0,28	[°]
mk	45		[g]
mk,ges	315		[g]
msw	230		[g]
ra	37		[mm]
ri	21		[mm]
h	10		[mm]
rho	7.9		[g/cm³]
V	29154		[mm³]
R fr,ring	30		
R f(ra,ri)	29		
sin2(alpha	0,5299		
tan(alpha)	0,2867		

beta	i		
beta	1	0,0	0,00
beta	2	51,4	0,90
beta	3	102,9	1,80
beta	4	154,3	2,69
beta	5	205,7	3,59
beta	6	257,1	4,49
beta	7	308,6	5,39
z"	i		
z"	1	569,9	
z"	2	355,4	
z"	3	-126,8	
z"	4	-513,5	
z"	5	-513,5	
z"	6	-126,8	
z"	7	355,4	
Fmi	i		
Fmi	1	25,6	
Fmi	2	16,0	
Fmi	3	-5,7	
Fmi	4	-23,1	
Fmi	5	-23,1	
Fmi	6	-5,7	
Fmi	7	16,0	
M(Fmi)	i		
M(Fmi)	1	0,74	
M(Fmi)	2	0,29	
M(Fmi)	3	0,04	
M(Fmi)	4	0,60	
M(Fmi)	5	0,60	
M(Fmi)	6	0,04	
M(Fmi)	7	0,29	

Jz	208436
(Jx =) Jy	106137
Jyz	27105
omega	262
Jy/mk,ges	337
Jy/msw	461
Jz/mk,ges	662
Jz/msw	905
msw/mk,ges	0,73

n	2500	[1/min]
alpha	16	[°]

Mk,ges	2,6032
Msw	1,8578

It can be seen that the influence of the piston masses predominates, resulting in the up-regulation behaviour of the swash plate or tilt plate with increasing speed of rotation.

In this case, therefore, $M_{k,ges} > M_{SW}$.

Fig. 3 shows a diagram for an almost identical drive mechanism, which diagram results from the following calculation scheme, α being varied from 0° to 16° in this case too:

Determination of swash plate tilt moment

theta	0	0,00	[°]
n (p)	7		-
R	29		[mm]
n	2500		[1/min]
alpha	16	0,28	[°]
mk	45		[g]
mk,ges	315		[g]
msw	415		[g]
ra	37		[mm]
ri	21		[mm]
h	18		[mm]
rho	7,9		[g/cm³]
V	52477		[mm³]

beta	i		
beta	1	0,0	0,00
beta	2	51,4	0,90
beta	3	102,9	1,80
beta	4	154,3	2,69
beta	5	205,7	3,59
beta	6	257,1	4,49
beta	7	308,6	5,39

z*	i	
z*	1	569,9
z*	2	355,4
z*	3	-126,8
z*	4	-513,5
z*	5	-513,5
z*	6	-126,8
z*	7	355,4

Jz	375185
(Jx =) Jy	198786
Jyz	46739
omega	262
Jy/mk,ges	631
Jy/msw	480
Jz/mk,ges	1191
Jz/msw	905
msw/mk,ges	1,32

R fr,eing	30
R f(ra,ri)	29
sin2(alpha	0,5299
tan(alpha)	0,2867

Fmi	i	
Fmi	1	25,6
Fmi	2	16,0
Fmi	3	-5,7
Fmi	4	-23,1
Fmi	5	-23,1
Fmi	6	-5,7
Fmi	7	16,0

M(Fmi)	i	
M(Fmi)	1	0,74
M(Fmi)	2	0,29
M(Fmi)	3	0,04
M(Fmi)	4	0,60
M(Fmi)	5	0,60
M(Fmi)	6	0,04
M(Fmi)	7	0,29

n	2500	[1/min]				
alpha	16	[°]	Mk,ges	2,6032	Msw	3,2034

In this case, $M_{k,ges} < M_{SW}$.

This calculation scheme shows that, compared to the calculation relating to Fig. 2, the thickness or height of the swash plate or tilt plate was increased from 10 mm (Fig. 2) to 18 mm (Fig. 3). The consequence thereof is that the relevant mass moment of inertia J_z is increased to about twice its value in comparison. In Fig. 3, there can be seen a down-regulation behaviour of the tilt plate drive mechanism. This tendency is indicated by the arrow "n" in Fig. 3, "n" meaning the speed of rotation of the tilt plate and drive shaft. The arrow "n" in Fig. 2 has, of course, the same meaning, but in that case the arrow points in the opposite direction, which is intended to indicate up-regulation with increasing speed of rotation.

Figs. 2 and 3 reflect the prior art. In that context, the up-regulation behaviour corresponding to Fig. 2 is frequently to be found in current mass-produced R134a compressors. In the case of more recent developments, on the other hand, attempts are being made to change this tendency to the opposite, namely to that corresponding to Fig. 3.

Starting from the mentioned prior art, the problem of the present invention is to provide an axial piston compressor of the kind mentioned at the beginning wherein changes in the speed of rotation have a minimum influence on the tilt plate apparatus of the compressor, that is to say it is not the delivery output but rather the tilt angle of the tilt plate which should be influenced as little as possible by the speed of rotation.

The problem is solved in accordance with the invention by the characterising features of claim 1, advantageous developments and details of the invention being described in the subordinate claims.

The central concept of the present invention accordingly lies in matching the geometry and the dimensioning of the parts moved in translation, on the one hand, and of the parts moved in rotation, on the other hand, to one another so that the moments caused thereby are always of approximately equal magnitude so that the tilt plate tilt angle remains substantially constant in the case of changing speeds of rotation.

As a result, the following advantages are obtained:

- advantageous dynamic behaviour: reduced hunting and counter-regulation by valves;
- less spread between the characteristic curves; as a result, each operating point can be optimally considered during design and placed in the characteristic diagram (of particular interest in the case of CO_2 compressors because in the case of these compressors, in comparison to R134a compressors, HP (heat pump) operating points have to be taken into account in addition to AC (air-conditioning) operating points);
- by superimposing a moment due to the location of the centre of gravity it is possible to obtain, approximately, the adjusted plots of the characteristic curves and also to displace them in required manner.

The aim in accordance with the invention is accordingly to adjust the sum of the translational and rotary masses to "zero". From the above-mentioned equations for M_{SW} due to (moment of deviation) and $M_{k,ges}$ it emerges that when those two moments are equal the influence of the speed of rotation "ω^2" is cancelled out.

Furthermore, the inventors have recognised that it is nevertheless not possible to avoid an influence of the tilt angle of the tilt plate. This emerges from the plots of tan(alpha) and sin(2alpha). The plots of those angle functions are shown in Fig. 1. It can be deduced therefrom that the influence at a small tilt angle is very small but then increases considerably in the case of relatively large tilt angles. Usually, the tilt plate tilt angles are limited by a minimum value α_{min} and a maximum value α_{max}. It is possible to envisage limits of 0°, on the one hand, and 30°, on the other hand. In practice, the minimum and maximum values are between about 0.6° and 18°. On that basis, it can be seen from Fig. 1 that in the latter range, a deviation of about 13 % has to be expected. This means that when a balance between up-regulating and down-regulating masses has been achieved by constructional means for a minimum tilt angle, an undesirable tilting behaviour (speed of rotation influenced by the tilt angle) has to be expected at the opposite limit of the tilt plate.

In accordance with the present invention the drive mechanism should be so designed that, at least approximately, the mentioned undesirable behaviour with respect to change in the speed of rotation at different tilt angles is greatly reduced.

Fig. 4 shows the tilting characteristic of the tilt plate for a drive mechanism for which, at a tilt angle of 1°, the mass forces/moments have been so adjusted by constructional means that up-regulating and down-regulating tilt moments approximately counterbalance one another. The corresponding calculation scheme is as follows:

Determination of swash plate tilt moment

theta	0	0,00	[°]
n (p)	7		-
R	29		[mm]
n	2500		[1/min]
alpha	1	0,02	[°]
mk	45		[g]
mk,ges	315		[g]
msw	302		[g]
ra	37		[mm]
ri	21		[mm]
h	13,13		[mm]
rho	7,9		[g/cm³]
V	38279		[mm³]
R fr,eing	30		
R f(ra;ri)	29		
sin2(alpha	0,0349		
tan(alpha)	0,0175		

beta	i		
beta	1	0,0	0,00
beta	2	51,4	0,90
beta	3	102,9	1,80
beta	4	154,3	2,69
beta	5	205,7	3,59
beta	6	257,1	4,49
beta	7	308,6	5.39

z*	i	
z*	1	34,7
z*	2	21,6
z*	3	-7,7
z*	4	-31,3
z*	5	-31,3
z*	6	-7,7
z*	7	21,6

Fmi	i	
Fmi	1	1,6
Fmi	2	1,0
Fmi	3	-0,3
Fmi	4	-1,4
Fmi	5	-1,4
Fmi	6	-0,3
Fmi	7	1.0

M(Fmi)	i	
M(Fmi)	1	0,05
M(Fmi)	2	0,02
M(Fmi)	3	0,00
M(Fmi)	4	0.04
M(Fmi)	5	0,04
M(Fmi)	6	0,00
M(Fmi)	7	0.02

Jz	273677
(Jx =) Jy	141183
Jyz	2312
omega	262
Jy/mk,ges	448
Jy/msw	467
Jz/mk,ges	869
Jz/msw	905
msw/mk,ges	0,96

n	2500	[1/min]
alpha	1	[°]

Mk,ges	0,1585
Msw	0,1585

n	2500	[1/min]				
alpha	1	[°]	Mk,ges	0,1585	Msw	0,1585
n	2500	[1/min]				
alpha	8	[°]	Mk,ges	1,2759	Msw	1,2515
n	2500	[1/min]				
alpha	16	[°]	Mk,ges	2,6032	Msw	2,4061
n	11000	[1/min]				
alpha	1	[°]	Mk,ges	3,0679	Msw	3,0678
n	11000	[1/min]				
alpha	8	[°]	Mk,ges	24,7014	Msw	24,2296
n	11000	[1/min]				
alpha	16	[°]	Mk,ges	50,3983	Msw	46,5820

The tilting characteristic shown in Fig. 4 is based on a pressure of 120 bar on the high-pressure side and 35 bar on the suction pressure side at speeds of rotation of: 600 rpm, 1200 rpm, 2500 rpm, 5000 rpm, 8000 rpm and 11,000 rpm. In the above calculation scheme, the balance-sheet of moments has been calculated for a minimum tilt angle of 1° and adjusted by appropriate selection of the tilt plate geometry. For all further computations, the internal and external diameter of the tilt plate were left unchanged; only the height of the tilt plate is varied for adaptational purposes. In the context of the pistons, the mass has been set at a constant 45 g for all further calculations. To begin with, the centre of gravity of the tilt plate is located directly in the tilt-providing articulation of the tilt plate for the purpose of simplifying the balance-sheet of moments. The position of the pistons is governed approximately by the tilt plate geometry by way of the relation $(r_a+r_i)/2$. The numbers and details given are to be regarded as being used solely by way of example; qualitatively, though, the relationships are also representative of other assumptions. The example treated here relates to a compressor in a CO_2 application.

The diagram shown in Fig. 4 shows a regulation behaviour result at relatively large tilt angles which is not very satisfactory. The compressor exhibits a high degree of up-regulation solely on account of the tilt moments due to the pistons moved in translation. In view of the aim stated hereinbefore, this result is not exactly optimal. The balancing of the tilt moments at a minimum tilt angle of α = 1° is demonstrated by the fact that at that tilt angle $M_{k,ges}$ is almost equal to M_{SW}.

The diagram shown in Fig. 5 is based on the following calculation and results:

Determination of swash plate tilt moment

theta	0	0,00	[°]
n (p)	7		-
R	29		[mm]
n	2500		[1/min]
alpha	8	0,14	[°]
mk	45		[g]
mk,ges	315		[g]
msw	309		[g]
ra	37		[mm]
ri	21		[mm]
h	13.404		[mm]
rho	7,9		[g/cm³]
V	39078		[mm³]

R fr,sing	30
R f(ra;ri)	29
sin2(alpha	0,2756
tan(alpha)	0,1405

beta	i		
beta	1	0,0	0,00
beta	2	51,4	0,90
beta	3	102,9	1,80
beta	4	154,3	2,69
beta	5	205,7	3,59
beta	6	257,1	4,49
beta	7	308,6	5,39

z*	i	
z*	1	279,3
z*	2	174,2
z*	3	-62,2
z*	4	-251,7
z*	5	-251,7
z*	6	-62,2
z*	7	174,2

Fml	i	
Fml	1	12,6
Fml	2	7,8
Fml	3	-2,8
Fml	4	-11,3
Fml	5	-11,3
Fml	6	-2,8
Fml	7	7,8

M(Fml)	i	
M(Fml)	1	0,36
M(Fml)	2	0,14
M(Fml)	3	0,02
M(Fml)	4	0,30
M(Fml)	5	0,30
M(Fml)	6	0,02
M(Fml)	7	0,14

Jz	279388
(Jx =) Jy	144316
Jyz	18615
omega	262
Jy/mk,ges	458
Jy/msw	467
Jz/mk,ges	887
Jz/msw	905
msw/mk,ges	0,98

n	2500	[1/min]					
alpha	8	[°]	Mk,ges	1,2759		Msw	1,2759

n	2500	[1/min]				
alpha	1	[°]	Mk,ges	0,1585	Msw	0,1615
n	2500	[1/min]				
alpha	16	[°]	Mk,ges	2,6032	Msw	2,4529
n	2500	[1/min]				
alpha	8	[°]	Mk,ges	1,2759	Msw	1,2759
n	11000	[1/min]				
alpha	1	[°]	Mk,ges	3,0679	Msw	3,1275
n	11000	[1/min]				
alpha	8	[°]	Mk,ges	24,7014	Msw	24,7611
n	11000	[1/min]				
alpha	16	[°]	Mk,ges	50,3983	Msw	47,4884

From Fig. 5 there can be likewise be seen a similar behaviour to Fig. 4, although the up-regulating effect has been slightly reduced. In the calculation, the following was altered with respect to the calculation relating to Fig. 4:

As the tilt angle for setting the balance-sheet of the relevant moments to zero there has been selected a medium tilt angle of $\alpha = 8°$.

In order to modify the moment of inertia of the tilt accordingly, the height of the tilt plate was very slightly increased, more specifically from 13.130 mm to 13.404 mm.

The balance-sheet of the moments in the calculation scheme shows balancing of the moments at a tilt angle of 8°. It can also be seen that the individual characteristic curves in the region of relatively low speeds of rotation are very close to one another and even have a slightly down-regulating effect. The curves then have a point of intersection approximately at a tilt angle $\alpha = 8°$ and afterwards become separated again with up-regulation behaviour.

The invention and the aim of the present invention will now be described in greater detail with reference to the further Figures. In detail, the Figures show the following:

Fig. 6 shows the regulation behaviour of a tilt plate designed in accordance with the invention;

Fig. 7 shows the regulation behaviour of a tilt plate also designed in accordance with the invention but differently from Fig. 6;

to be corrected

Fig. 8 shows the regulation behaviour of a tilt plate according to the invention for a single speed of rotation of n = 5000 rpm with variation of the operating point (differently set high pressure and suction pressure);

Fig. 9 shows a preferred arrangement of a tilt ring drive mechanism for an axial piston compressor in a perspective view;

Fig. 10 X shows the tilt ring drive mechanism according to Fig. 9 in axial section along line II-II in Fig. ~~13~~;

Fig. 11 X shows the drive mechanism according to Fig~~s~~. 1 ~~and 2~~ in an exploded view;

Fig. 12 shows division of the drive mechanism according to Fig. 9 into four quadrants Q1, Q2, Q3 and Q4;

Fig. 13 shows, in diagrammatic manner, the co-ordinates of a tilt plate mechanism for calculation of the mass moment of inertia;

Figs. 14a - 21 show the influence of the location of the centre of gravity of the tilt plate relative to the tilt axis, the tilt axis defining the zero points for the y and z co-ordinates.

Fig. 6 and the following calculation scheme

Determination of swash plate tilt moment

theta	0	0,00	[°]
n (p)	7		-
R	29		[mm]
n	2500		[1/min]
alpha	16	0,28	[°]
mk	45		[g]
mk,ges	315		[g]
msw	329		[g]
ra	37		[mm]
ri	21		[mm]
h	14,292		[mm]
rho	7,9		[g/cm³]
V	41667		[mm³]
R fr,eing	30		
R f(ra;ri)	29		
sin2(alpha	0,5299		
tan(alpha)	0,2867		

beta	i		
beta	1	0,0	0,00
beta	2	51,4	0,90
beta	3	102,9	1,80
beta	4	154,3	2,69
beta	5	205,7	3.59
beta	6	257,1	4,49
beta	7	308.6	5.39

z*	i	
z*	1	569,9
z*	2	355,4
z*	3	-126,8
z*	4	-513,5
z*	5	-513,5
z*	6	-126,8
z*	7	355,4

Fmi	i	
Fmi	1	25,6
Fmi	2	16,0
Fmi	3	-5,7
Fmi	4	-23,1
Fmi	5	-23,1
Fmi	6	-5,7
Fmi	7	16.0

M(Fmi)	i	
M(Fmi)	1	0,74
M(Fmi)	2	0,29
M(Fmi)	3	0,04
M(Fmi)	4	0,60
M(Fmi)	5	0,60
M(Fmi)	6	0,04
M(Fmi)	7	0,29

Jz	297897
(Jx =) Jy	154552
Jyz	37981
omega	262
Jy/mk,ges	491
Jy/msw	470
Jz/mk,ges	946
Jz/msw	905
msw/mk,ges	1,04

n	2500	[1/min]
alpha	16	[°]

Mk,ges	2,6032
Msw	2,6032

n	2500	[1/min]				
alpha	1	[°]	Mk,ges	0,1585	Msw	0,1714
n	2500	[1/min]				
alpha	8	[°]	Mk,ges	1,2759	Msw	1,3540
n	2500	[1/min]				
alpha	16	[°]	Mk,ges	2,6032	Msw	2,6032
n	11000	[1/min]				
alpha	1	[°]	Mk,ges	3,0679	Msw	3,3191
n	11000	[1/min]				
alpha	8	[°]	Mk,ges	24,7014	Msw	26,2141
n	11000	[1/min]				
alpha	16	[°]	Mk,ges	50,3983	Msw	50,3972

show the regulation behaviour of a tilt ring drive mechanism which has been so dimensioned that moment balancing occurs at a tilt angle of 16°, wherein the tilt angle of 16° should be equal to α_{max}. The height of the tilt plate was adjusted to 14.292 mm. It should be mentioned at this point that other tilt plate parameters can also, of course, be used for adjusting the mass moment of inertia. The parameter of "tilt plate height" was, however, selected merely by way of example in order to provide a ready means of comparison.

The regulation behaviour of the tilt plate according to Fig. 6 shows an especially desirable result. In the entire operating range of the tilt plate, that is to say from α_{min} to α_{max}, there is no appreciable up-regulating effect of moments of moved masses; a wide spread of the characteristic curves is avoided, more specifically as a result of the fact that there is a point of intersection or convergence of the characteristic curves at the maximum tilt angle, that is to say α_{max}.

In the region between a medium tilt angle and the maximum tilt angle, especially at the maximum tilt angle, the sum of moments of the up-regulating and down-regulating mass forces is approximately zero.

At this point it should also be mentioned that the present description relates solely to the mass forces, inertia forces and resulting moments which influence tilting of the tilt plate and are due to the tilt plate and pistons and possible additional components. In the present case there is used a drive mechanism which has only a few component parts. Starting from the example on which this description is based, having a tilt plate, sliding blocks and pistons, more complex designs can also be envisaged, of course, for example a wobble plate compressor. In principle, that which is said here also holds true for those more complex arrangements.

In addition, the balancing of moments described here can be checked without its being necessary to put the compressor into operation. Starting from the above-mentioned basic equation for the balancing of moments it is clear that the characteristic curves can be calculated from the measured geometry and the measured component masses. Where applicable, the centre of gravity of the tilt plate also has to be determined as well.

Figs. 7 and 8 show a variant of the arrangement according to Fig. 6. Above all, Figs. 7 and 8 show that balancing of moments is also possible for a virtual tilt angle, for example a tilt angle whose value is above α_{max}. When, for example, the operating range of the tilt plate tilt angle extends from 1° to 16°, the drive mechanism can be so designed that the sum of the moments is zero at a virtual tilt angle of 22°. Fig. 7 shows that in such a case there is also obtained a regulation behaviour which comes very close to the desired goal. In Fig. 7, all the characteristic curves, more specifically the characteristic curves for different speeds of rotation, converge at a virtual tilt angle α = 22°.

Fig. 8 shows the characteristic curves for a single speed of rotation n = 5000 rpm but at different operating points, which are characterised by different ratios of high pressure and suction pressure, the high pressure being between 65 bar and 120 bar and the suction pressure between 25 bar and 50 bar. From Fig. 8 it can be seen that, depending on the operating point, the characteristic curve tends to be located in the curve diagram at relatively high pressures or at somewhat lower ones. In all the graphs, the characteristic curves have been calculated using a spring constant, in respect of the restoring spring (restoring in the direction of minimum stroke), of 60 N/mm. If the spring constant selected were to be lower, the curves would drop off less towards higher tilt angles. If the spring constant selected were to be higher, the curves would drop off more towards higher tilt angles.

Each of the curves for 5000 rpm is to be considered as being representative of a group of curves for a particular operating point. When it is taken into account that a particular regulation pressure of at least 2-3 bar above the suction pressure is required, an advantageous regulation behaviour is achieved by the plots having a certain slope which is linear as far as possible over a wide range. As a result it becomes clear that, for all operating ranges, a group of curves located close to one another is located more in the

desired region of the plot diagram than regulation curves which drift apart from one another to a relatively great extent, as is shown in Figs. 2 and 3.

In such negative cases, for example in the case of up-regulation behaviour in accordance with Fig. 2, it can be seen that parts of regulation curves can very readily lie in regions below the suction pressure (which is of course undesirable - in that case, the compressor cannot be operated at maximum stroke). In order to avoid that, weaker restoring springs would have to be used. However, it is then possible for an increasing characteristic to occur, especially at high speeds of rotation, in contrast to the falling characteristic at low speeds of rotation. Experience has shown that in such cases highly undesirable regulatory effects come about. In particular, a defined tilt angle can no longer be assigned to individual pressures (occurrence of relative maxima or minima and/or no or inadequate slope or inflection points).

Figs. 9 - 12 show a preferred arrangement of an advantageous tilt plate or tilt ring mechanism. This mechanism, or the corresponding tilt ring drive mechanism, is referred to by reference numeral 100. It comprises a tilt plate, annular in the present case, or tilt ring 107, which is modifiable in terms of its inclination relative to a drive shaft 104 and which is driven in rotation by the drive shaft, the tilt ring being in articulated connection both with a sliding sleeve 108 mounted on the drive shaft 104 so as to be axially displaceable and also with a supporting element 109, which is spaced away from the drive shaft 104 and rotates together with the latter. This articulated connection is in the form of axial support, as can be seen especially well from Figs. 10 and 11. The co-operation of the tilt ring 107 with the axial pistons corresponds to that which is in accordance with the prior art according to DE 197 49 727 A1.

The pivotal mounting of the tilt ring 107 defines a pivot axis 101 extending in a transverse direction to the drive shaft 104. The pivot axis is defined, moreover, by two mounting pins 102, 103 mounted coaxially on both sides of the sliding sleeve 108 (see Fig. 11). The mounting pins 102, 103 are mounted in radial bores in the tilt ring 107. These radial bores are referred to in Fig. 11 by reference numeral 130. For the purpose, the sliding sleeve 108 can additionally have, on both sides, mounting sleeves 105, 106 (see Fig. 11), which bridge the annular space 119 between the sliding sleeve 108 and the tilt ring 107.

Of significance is the axial support of the tilt ring at the supporting element 109 arranged to rotate together with the drive shaft 104. That support is provided by means of a supporting arc 110 in operative connection with the tilt ring 107. The supporting arc 110 is so constructed that it overlaps an articulated arrangement effective between the pistons and tilt ring, more particularly in such a manner that, irrespective of the inclination of the tilt ring 107, the possibility is excluded of a collision between the tilt ring 107 and supporting arc 110, on the one hand, and a piston foot 111 surrounding an articulated arrangement, on the other hand (see Fig. 10 in that respect). The piston associated with the piston foot 111 is referred to by reference numeral 118. The supporting element 109 is part of a disc 112 rotating together with the drive shaft 104.

The supporting surface of the arc 110 extends approximately concentrically to the centre-point of the articulated arrangement effective between the pistons 118 and tilt ring 107. The axial support is accordingly effective outside the afore-mentioned articulated arrangement, with the consequence that the articulated arrangement that is effective between the pistons and tilt ring is not impaired by axial support measures. This is valid especially for the dimensioning of the afore-mentioned articulated arrangement. As in the prior art, the articulated arrangement is defined by two articulating blocks 121, 122 in the shape of segments of a sphere (see Fig. 10), between which the tilt ring 107 slidingly engages. Associated with the spherical bearing surfaces of the articulating blocks 121, 122 are corresponding spherical recesses on those end faces of the piston foot 111 which face one another.

Furthermore, it can be seen that, in the arrangement shown, the pivotal mounting of the tilt ring 107 serves only for transmitting torque and the supporting element 109 serves only for axially supporting the pistons 118 and/or for providing support against gas forces. The transmission of torque is accordingly de-coupled from the axial support of the tilt ring 107.

Also of particular interest is the supporting surface for the supporting arc 110 on the supporting element 109. That supporting surface is in the form of an arcuate or cylindrical bearing surface 123. In order to avoid displacement of the supporting line, when the inclination of the tilt ring 107 changes, that is to say displacement away from the centre of the pistons 118, the supporting arc 110 is mounted so as to be displaceable in a radial direction relative to the tilt ring 107.

In other respects, reference is made, regarding the design of this drive mechanism, to the German Patent Application No. 103 35 159.0 filed by the Applicant.

In accordance with Fig. 12, the described drive mechanism has been divided into four regions or quadrants Q1, Q2, Q3 and Q4 with respect to the drive shaft 104 and the tilt ring 107. The first quadrant Q1 is bounded by the drive shaft 104 and the front face of the tilt ring 107 including the piston support, that is to say the face facing the piston. The other quadrants follow on anti-clockwise from quadrant Q1 around the pivot or tilt axis of the tilt ring 107.

In the examples hitherto, it has been assumed that the centre of gravity of the tilt ring 107 coincides substantially with the tilt axis, which extends perpendicular to the mid-axis of the drive shaft. Following on therefrom, however, compressors according to the prior art frequently have centres of gravity in the region of the tilt plate where the centre of gravity of the tilt plate does not coincide with the tilt axis. In according with the invention it is possible to envisage the deliberate incorporation of an "offset". In that case, centres of gravity in the quadrants Q have the following effects:
Q1 (positive z and y co-ordinates): down-regulating
Q3 (negative z and y co-ordinates): down-regulating
Q2 (positive z and negative y co-ordinates): up-regulating
Q4 (negative z and positive y co-ordinates): up-regulating

The z co-ordinate extends parallel to the mid-axis of the drive shaft, preferably along it. The y co-ordinate extends perpendicular thereto.

If no additional compensation of masses is provided, the centre of gravity in the case of arrangements according to the prior art accordingly lies, very frequently, in the fourth quadrant Q4.

It is of course possible that a centre of gravity arranged in any particular quadrant will change shaft side, relative to the mid-axis of the shaft, when the swash plate tilts, with the consequence, for example, that up-regulation behaviour will be transformed into down-regulation behaviour. It has, however, also been found of course that in the region

of the shaft axis the centrifugal force and a tilting moment possibly arising therefrom tend to keep themselves within limits.

Overall, with respect to the two moments already mentioned $M_{k,ges}$ and M_{SW}, which can be compensated for certain tilt angles, a further tilt moment due to centrifugal force comes into effect, which is included as a component in M_{SW} or msw, which is included in the moment of deviation J_{yz}, which is in turn included in M_{SW} ($M_{SW} = J_{yz} \cdot \omega^2$).

Hereinbelow there will also be described particular arrangements relating to the location of the centre of gravity of the tilt plate:

An "offset" is provided which has an up-regulating effect. This means that the centre of gravity of the tilt plate is located either in the second quadrant or in the fourth, that is to say in Q2 or Q4.

Various cases are shown in Figs. 14a to 19.

Fig. 14a shows the regulation behaviour for a centre of gravity which coincides with the tilt axis. In Fig. 15a, the centre of gravity has been displaced, more specifically either into the second quadrant or into the fourth with the co-ordinates y=3/z=-2 or y=-3/z=2, respectively. The consequence of this displacement of the centre of gravity is that an additional (partial) moment is produced which has an up-regulating effect. In the case of the selected centre of gravity, the variation in moment in the tilt angle range of from, for example, 1° to 16° is not so great that it is possible to say that the characteristic curves can be shifted in parallel to greater pressures, the higher the speed of rotation. In principle it is possible, as a result, to bring the curves somewhat closer together in a narrow band. According to Fig. 1, for large tilt angles, however, there is obtained a balanced moment balance-sheet, approximately, and as a result a small amount of variability (about 10-16 % swash plate tilt angle). In the region of relatively small and medium tilt angles, but especially at small tilt angles, an up-regulating moment is obtained. The up-regulating tilt moment in the low speed of rotation range can be attractive because it is desirable to adjust the tilt angle in connection with, for example, coupling-free operation in the "Off" operating state of the air-conditioning system for power consumption that is as low as possible. However, limits are set on minimisation of the tilt angle because the compressor, for regulation to a greater stroke, must always be

to be corrected

capable of building up a small pressure difference. If the tilt angle and, as a result, the stroke have been set too low, the compressor has difficulty in up-regulating. In that case, the dynamics can help in adjusting the compressor to a greater stroke. In Figs. 14b and 15b, the influence of the location of the centre of gravity is shown by way of comparison. It is seen as an advantage that, as a result of the feature, the characteristic curves can be shifted approximately in parallel to greater pressures. It is desirable that the compressor down-regulates in the relatively high speed of rotation range and up-regulates in the relatively low speed of rotation range. This is shown in Fig. 15a by the arrows "n" pointing in opposite directions. Figs. 14b and 15b correspond to Figs. 14a and 15~~b~~, more specifically in each case in the region of small tilt angles and to an enlarged scale.

Because of the gas process in real terms and the clearance volumes, which cannot be avoided and which are highly relevant during compression, a kind of maximum or plateau comes about in the region of low tilt angles which does not allow a clear association between pressure and speed of rotation and frequently results in regulation problems.

As a result of the present invention, that influence can be reduced, that is to say the slope of the characteristic curves tends to be somewhat increased.

In Fig. 15b, the speeds of rotation are staggered from bottom to top as follows: 600 rpm, 1200 rpm, 2500 rpm, 5000 rpm, 8000 rpm and 11,000 rpm, the characteristic curves for 600 rpm and 1200 rpm almost coinciding. When, at the minimum tilt angle, the speed of rotation is increased from 600 rpm or 1200 rpm to about 2500 rpm, the tilt angle increases to 2.5°.

In the case of current designs, attempts are being directed, as already mentioned, at minimising the minimum tilt angle. Accordingly, for example, the minimum tilt angle can be only about 0.6°. Such a range can, however, be very critical for problem-free compressor start-up. An increase in the minimum tilt angle due to an increase in the speed of rotation even by a few tenths of a degree can therefore be very useful.

Figs. 16 to 19 show further examples of differing centre of gravity locations, from which the following findings are derived:

Depending on the tilt angle, the drive mechanism provides up-regulating or down-regulation behaviour.

In some regions, the characteristic curves are very close together or even lie on top one one another.

The characteristic curves always intersect at one point.

With reference to Figs. 20 and 21, the case is also described by way of example where a moment additionally brought about by an appropriate centre of gravity location has a down-regulating effect. The centre of gravity is located either in the first quadrant or in the third. As the speed of rotation increases, the characteristic curves are shifted to lower pressures as a result of such a centre of gravity location. Such dimensioning of the components can be advantageous in order to bring about down-regulation, in the case of increasing compressor speed of rotation, solely as a result of the sum of the active moments.

At first sight, the regulating characteristic shown has nothing to do with compensation of mass forces because at a low speed of rotation the tilt angle is kept constant. Consequently, to begin with, the compensation of mass forces relates only to the balancing of the moments $M_{k,ges}$ and M_{SW} at a predetermined tilt angle. The described case of "down-regulating moment due to the location of the centre of gravity" also falls within that category.

Following on therefrom, the invention relates to balancing the moments $M_{k,ges}$ and M_{SW} and also an additional moment due to the location of the centre of gravity (up-regulating).

To summarise once again, the advantage of the described examples of embodiments compared to the prior art according to, for example Fig. 3, lies substantially in the fact that there is one tilt angle at which balancing of the moments occurs.

As a result, the speed of rotation of the drive mechanism then has only a moderate influence, if any influence at all, on the characteristic curves associated with the

individual speeds of rotation, insofar as the angle functions tan(α) and sin(2α) have an appreciable effect. This effect is, in any case, small for small "α".

In addition, as a result of the location of the centre of gravity, a different behaviour can be set. Overall, however, the influence of the speed of rotation is considerably reduced.

As a result of the close grouping of the characteristic curves associated with the speeds of rotation in the case of many operating points, dimensioning of, for example, the restoring spring in desired manner is more simple.

Fig. 13 is intended to serve for better understanding of the formulae mentioned at the beginning for calculation of the tilt moments in question herein. In that regard, reference is made to the corresponding formulae and the above explanation of the individual variables.

With reference to the above-described characteristic and regulation curves it should be pointed out again that for the characteristic of the compressor according to the invention it is of very major importance and also recognition that the characteristic or regulation curves for differing speeds of rotation run approximately parallel to one another. In the case of a characteristic curve plot of such a kind, the conditions according to the invention are met.

Furthermore, it should be pointed out that the tilt angle of the tilt plate or tilt ring 107 changes by about 2° to 4° in the case of an increase in the speed of rotation from minimum to maximum, more particularly especially under the condition of an approximately constant pressure in the drive mechanism chamber. This change means that the tilt angle is practically constant under the given conditions. Likewise, the piston stroke is then also practically constant.

The spring constant of the restoring spring acting on the tilt plate is generally optimised for a particular speed of rotation. In the case of substantial variations in the speed of rotation, this has a disadvantageous effect in the case of conventional compressors, more specifically because of the relatively large spread in the regulatory characteristic curves between minimum and maximum speed of rotation.

In accordance with the invention, however, the regulatory characteristic curves are located very close to one another and run almost parallel to one another. Accordingly, an optimum spring constant can be set for a relatively close group of characteristic curves with the consequence that the spring constant is set, or can be set, almost optimally for all speeds of rotation between the minimum and maximum speeds of rotation. The spring constant is about 40 to 90 N/mm, especially about 40 to 70 N/mm.

Finally it should also be pointed out that, in the calculation of the mass moments of inertia described hereinabove, especially that of the moment of deviation, there should preferably also be taken into account a so-called Steiner component ($y_s \cdot z_s \cdot m$), the moment of deviation being described as follows when this is taken into account:

$$J_{\overline{YZ}} = J_{YZ} + y_s \cdot z_s \cdot m$$

For small tilt angles of the tilt plate, for example of a tilt ring, the J_{YZ} component is smaller than the Steiner component $y_s \cdot z_s \cdot m$. The component J_{YZ} always down-regulates the compressor whereas the component $y_s \cdot z_s \cdot m$ always up-regulates (set by the position in the quadrant Q2, Q4 according to Fig. 12).

From the above-mentioned considerations it emerges that the moment of deviation has two components having contrary influences, that is to say both an up-regulating and a down-regulating component.

The components in question come into effect in each case after a threshold tilt angle α_G has been exceeded, the following applying for $\alpha < \alpha_G$:

$y_s \cdot z_s \cdot m > J_{YZ}$ (up-regulating),

and the reverse for $\alpha > \alpha_G$ (that is to say, down-regulating).

All features disclosed in the application documents are claimed as being important to the invention insofar as they are novel on their own or in combination compared with the prior art.